Theoretical Evaluation of Flow through a Mixed flow Compressor Stage

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Abstract

Flow through mixed flow compressor stage was evaluated theoretically to understand internal flow behaviour. Various shroud configurations in terms of clearance has been considered to see the effect of clearance on compressor performance. The blade to blade relative flow variation from impeller inlet to outlet was evaluated through theoretical analysis. A 3D viscous commercial code Fluent was used to analyse the flow through the impeller channel. Computations have been carried out for different operating conditions of speed and mass flow rates covering a wide range from choke to surge. It is identified constant tip clearance gives better performance over variable tip clearance. The recirculation zone at inlet of the impeller near the tip gets vanished as the flow reaches impeller exit. The classical jet wake flow at impeller exit as seen in radial machines was absent for the mixed flow impeller.

Nomenclature

C Absolute velocity (m/s)
k Turbulence kinetic energy(m^2/s^2)
m Mass flow rate (kg/s)
N Rotational speed (RPM)
P Pressure (Pascal)
PS Pressure surface
SS Suction surface
t Time, tip ,blade-to-blade pitch
T Temperature (K)
U Blade tip speed (m/s)
X, Y, Z Cartesian co-ordinates
R, θ, Z Cylindrical co-ordinates
y Distance from suction surface towards pressure surface
η Adiabatic efficiency (%)
ε Dissipation rate (m^2/s^3)
θ_0 Temperature correction factor for standard condition=T_{o1}/T_{atm}
δ Pressure correction factor for standard condition =P_{01}/P_{atm}
z Distance from hub to shroud in

Subscripts

atm Ambient
d Design
avg averaged property
tan Tangential
0 Stagnation state
1 Impeller inlet
2 Impeller outlet

Introduction

Compressors for conventional short range missiles with small turbojet or turbofan engines and those for helicopters require moderate pressure ratios and mass flow rates. A small overall size is a critical requirement. The problem of large frontal area associated with radial compressors can be solved by using a mixed flow impeller with a split diffuser. The design of mixed flow compressors has not developed as comprehensively as axial compressors due to structural limitations, limited computational capabilities and limited experimental data base in the past. But in recent years the design of mixed flow compressors has developed many folds due to rapid progress in computational techniques, computational power, manufacturing capabilities and better understanding of the flow physics. This paper describes the theoretical flow analysis of a mixed flow compressor stage with backswept and leaned impeller configured for a small gas turbine engine. The aerodynamic design with appropriate meridional and blade shaping through analytic functions were used to design the mixed flow compressor stage.

Most of the pioneering works in the design and analysis of mixed flow compressors started in 1940s and 50s work presented by Austin King et al[1]. In 1942, was probably one of the earliest works regarding mixed flow compressors. After some years, Laskin and Kofskey[2] investigated a mixed flow compressor stage with vane-less diffuser. In the same year, 1947, Barina [3] preformed experiments on two vanes-less diffusers for mixed flow compressors with different rates of passage curvature. The first document for the design of mixed flow compressor was that described by Goldstein[4]. The pioneering work in meridional plane analysis for mixed flow impellers was done by Hamrick[5]. Stanitz[6] described a method for design and analysis of one-dimensional compressible flow in vane-less diffusers of radial and mixed flow compressors by considering friction heat transfer and area change. Dallenbach[7] presented a method for aerodynamic design of centrifugal and mixed flow compressors to achieve prescribed impeller blade loading distribution for the impellers with radial bladed elements. In 1975, Wallace[8] et al presented computer-aided design procedure for radial and mixed flow compressors. Sakai et al[9] conducted experimental study on vane-less diffusers for mixed flow machines. Work done by Sakai and Wallace et al is very useful for mixed flow diffuser design. Musgrave and Plehn[10] presented a design and test results for a mixed flow compressor with a pressure ratio of 3:1. Awai et al[11] studied axially curved mixed flow vane-less diffusers. Niizeki et al[12] presented a design method to produce a small static pressure gradient normal to passage for radial curve mixed flow vane-less diffusers. Flow measurements showed high performance could be obtained using this method. Zangeneh[13] presented a compressible fully three dimensional inverse design method for radial and mixed flow turbo machinery blades. Monig et al[14] experimentally investigated a rotor designed for a 5:1 pressure ratio mixed flow supersonic compressor stage. Eisenlohr and Benfer([15] investigated a mixed flow compressor stage and also presented design steps for the mixed flow compressor. Zangench et al([16] presented a numerical method for suppression of secondary flows in centrifugal and mixed flow impellers. It can be observed that from 1980 to till date many attempts have been made for a successful mixed flow compressor design and there has been a rapid progress in the field, however very few successful stage designs are found in literature.

Aerodynamic design

The frontal area is one of the design parameter for small air-craft engine for good overall performance of the engine. Minimum frontal area is needed to reduce the engine size. This advantage has been obtained by designing a mixed flow compressor stage. The design of mixed flow compressor stage was based on centrifugal compressor design methodology.

The aerodynamic design was carried out for given super charge inlet condition. The designed mixed flow impeller has 11 full length blades and 11 splitter blades running at 39840 rpm. The designed relative inlet Mach number is 1.1 and hub tip ratio is 0.4 and the impeller outlet is inclined. The exit cone angle is 60°. Empirical correlation based on experimental results was used to calculate the stagnation pressure loss in impeller and diffuser. The designed stagnation pressure ratio is 4.55 with total to total adiabatic efficiency of 94%. The static to total stage efficiency is 80%. Different diffuser configuration were tried and optimised. The final optimised vane diffuser has 14 vanes with hub diameter 358mm and tip diameter 373mm. The impeller and diffuser coordinates were generated using analytical function and they were used as an input for CNC machining of the components. The same coordinates were also used for CFD analysis.

Geometric modelling and meshing

Figure-1 show the CAD model of compressor stage designed and generated in solid works CAD package. As the domain is axi-symmetric, a single channel consisting of one main blade and one splitter blade is considered for flow analysis. Impeller domain is generated using Gambit turbo tool pre-processor. Turbo tool was used for constant tip clearance configuration. Whereas for variable tip clearance blade to blade domain is constructed using Gambit. The flow domain is divided into multiple volumes to form cube like structure so as to fit hexahedral element. Meshing was carried out using Gambit. Hexahedral mesh elements were used in all volumes except near the clearance region over the main blade and splitter blade. Grid independence study was carried out to get optimum grid size. For the present analysis there are about 3 lakh elements.

Computational Methodology

For the present work, computations were performed using commercial code namely Fluent. A 3D viscous flow through impeller was analyzed. Segregated solver is used. The numerical fluxes are estimated with first order upwind scheme. The continuity equation, which governs mass conservation, is used to determine the pressure field in the pressure-based method. Turbulence is modeled using standard k-ε model. When the integrated residual of the pressure correction equation was reduced by four orders of magnitude from its initial value and the integrated residuals of all the variables were reduced by six orders of magnitude, the solution was considered as converged. The mass flow summary, the residual history of the variables and the monitor point values were used to judge the convergence behavior of the theoretical analyzed results. The mass imbalance was checked after the solution was converged. In all the cases the observed imbalance in mass flow rate was as low as 0.001%.

Boundary Conditions:

Figure-2 shows the flow domain considered. This figure also shows the boundary conditions assigned in the present analysis. Inlet boundary conditions used were total pressure, total temperature, and absolute flow angle. The distribution of these quantities was assumed constant throughout inlet boundary and hence single value was given. At exit, static pressure (sometimes termed as backpressure) was specified. Periodic boundary conditions were imposed on side faces of tip clearance, upstream and downstream of the impeller blade. The shroud casing was considered stationary. Hub surface and blades of impeller were specified as rotating.

Results and discussions

The flow analysis through the mixed flow compressor was carried out to understand the flow behaviour near the shroud for different tip clearances. Two configurations of tip clearance were considered. In the first configuration a constant tip clearance in which the magnitude of tip clearance (gap between the blade and casing) from inlet to exit of the impeller remains constant. In this configuration % of tip clearance in terms of blade height varies from inlet to exit. In the second configuration the tip clearance is made to vary. Here the tip gap varies from inlet to exit of the impeller. For this configuration the % of tip clearance in terms of local blade height remains constant.

Analysis was carried out for three constant magnitude of tip clearance namely 0.4mm, 0.5mm and 0.6mm. For variable tip clearance the percentage clearance was maintained at 2%, 3% and 4%. Both cases are studied under three different operating speeds namely 90%, 100% and 110% of design speed.

Figure-4 shows the performance characteristic of the impeller for constant tip clearance for different operating speeds. In this figure the adiabatic efficiency plot is also shown. Figure-5 shows the performance of the impeller for variable tip clearance. It is observed from these figures, for a given configuration the increase in tip clearance reduces the impeller pressure ratio as well as adiabatic efficiency. The efficiency for a given speed reaches a maximum value at particular mass flow rate which is close to the design flow coefficient of the impeller. By comparing the Figure-4 and Figure-5, it is observed that constant tip clearance provides better performance than a variable tip clearance in terms of pressure ratio and efficiency. This is because in variable clearance the magnitude of clearance between blade and shroud at inlet is large and large cross flow exists between pressure to suction surface of the blade. It is also observed at given speed in variable tip clearance the drop in pressure ratio is much higher as compared to constant tip clearance. It is clear from the Figure-5, the variable tip clearance provides larger swallowing capacity and choking limit is increased. The choke mass flow value does not change in case of constant tip clearance as compared to variable tip clearance.

The peak pressure ratio achieved was obtained from the previous performance graphs. This peak pressure ratio is compared with zero tip clearance at design point. Drop in peak pressure was calculated.

Figure-6 shows the percentage drop in pressure ratio against tip clearance. It is clear from this figure the stagnation pressure ratio drop is higher in variable tip clearance than in constant tip clearance. Similarly at each operating point the total to total adiabatic efficiency is calculated by mass averaged flow properties using

$$\eta = \frac{\left[\left\{ (P_{02})_{mavg} \Big/ (P_{01})_{mavg} \right\}^{\frac{\gamma-1}{\gamma}} - 1\right]}{\left[\left\{ (T_{02})_{mavg} \Big/ (T_{01})_{mavg} \right\} - 1\right]}$$

The drop in peak efficiency is evaluated from the performance graph. This is plotted against tip clearance for the design speed and is shown in Figure-7. The drop in total to total adiabatic efficiency in calculated using

$$\Delta\eta = \left[(\eta)_{without\ tip\ clearance} - (\eta)_{with\ tip\ clearance}\right]$$

In the case of variable tip clearance the drop in efficiency increases with tip clearance as compared to constant tip clearance. At higher tip clearances the rate of drop in efficiency reduces in case of variable tip clearance where as the rate of drop in efficiency remains constant at all tip clearances in case of constant tip clearance.

From the theoretical analysis velocity vector diagram at impeller outlet was evaluated. From the impeller outlet velocity diagram the work output in terms of change in tangential velocity component of the axial velocity was obtained. At design mass flow and design speed the change in work output in terms of percentage for two configurations of tip clearance was obtained using

$$\Delta\left(C_{\tan 2}\Big/U_2\right)(in\%) = \left[\frac{\left(C_{\tan 2}\Big/U_2\right)_{without\ tip\ clearance} - \left(C_{\tan 2}\Big/U_2\right)_{with\ tip\ clearance}}{\left(C_{\tan 2}\Big/U_2\right)_{without\ tip\ clearance}}\right] \times 100$$

Percentage in impeller work output is plotted against tip clearance in Figure-8. Due to drop in pressure ratio and efficiency with increase in tip clearance, the work output of impeller exit reduces with increase in tip clearance. The drop in impeller work output is higher in variable tip clearance as compared constant tip clearance. In overall the tip gap between rotating impeller and stationery shroud plays an important role in stage pressure ratio and efficiency. It is recommended to have a constant tip gap between impeller and stationery shroud. It is also observed that there was no optimum tip clearance obtained from the analysis. It suggests that further analysis need to be carried out to locate optimum tip clearance.

Figure-9 shows the static pressure normalised with reference to inlet pressure along the meridional distance. This figure indicates there is continuous rise in static pressure from FU to FD for both the configurations. Static pressure for constant tip clearance is more or less same for all clearances, where as variable tip clearances the total static pressure rise drops with tip clearances. It is also observed in case of constant tip clearance there is small drop in static pressure close to leading edge of the blade. Static pressure on suction and pressure surface near the tip of the blade along the blade was theoretically estimated for both the configuration. The variation of static pressure along blade surface with reference to normalised distance is shown in Figure-10. The area inside the curve represents blade loading. The static pressure variation on suction and pressure surface on splitter blade is also shown in the same figure. It is observed that, except near the leading edge of the splitter blade the static pressure on the splitter blade surface closely matches with the main blade surface. It is also observed that the static pressure variation at the tip is not greatly affected by tip gap for constant tip clearance except very close to the leading and trailing edge of blade. In the case of variable tip clearance local magnitude of static pressure on the suction and pressure surface reduces with increase in tip clearance. However the effective area within the pressure distribution curve remains same for all tip clearances indicating the blade loading marginally changes with tip clearance.

To study the flow behaviour in the meridional direction there are five planes taken within the impeller channel, which are shown in Figure-3. The relative velocity contours along the meridional planes for two configurations are shown in Figure-11. The relative velocity contours is shown for each plane for different tip clearance. It was observed that some where middle of the blade channel namely plane 2 there is a recirculation at tip in between blades. This recirculation zone increases in size as the tip clearance increases. This is observed close to pressure surface. In case of variable tip clearance the region of recirculation zone is higher as the flow reaches leading edge of splitter blade which is seen at Plane-III. As the flow moves further the recirculation zone vanishes and tip leakage flow exists from pressure to suction surface at the exit.

Figure-12 gives the blade to blade relative velocity variations at different meridional planes for three different clearances. The flow behaves like a potential flow. The relative velocity decreases from inlet to outlet of the impeller. Except at the shroud, the flow is uniform from blade to blade at impeller outlet without indicating classical jet wake as in the case of radial machines.

Conclusions

Constant tip clearance gives better performance over variable tip clearance. The change in operating range is more for variable tip clearance as compared to constant tip clearance. At a given speed effect of variable tip clearance on pressure ratio is larger. There is continuous rise in static pressure from inlet to outlet and effect of constant tip clearance on static pressure rise is marginal as compared with variable tip clearance. The same is observed in the blade loading distribution. Relative velocity contours shows recirculation zone near tip at inlet, moving towards splitter blade pressure surface as the flow reaches splitter blade leading edge. Only tip leakage flow is observed at exit. The flow at impeller exit is uniform without jet-wake.

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Figure-1 CAD model of mixed flow compressor

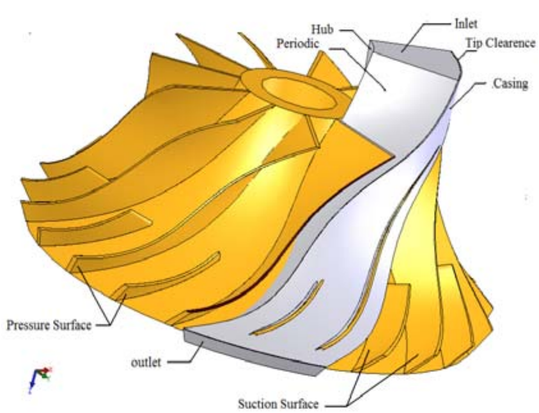


Figure-2 Flow domain with boundary conditions.

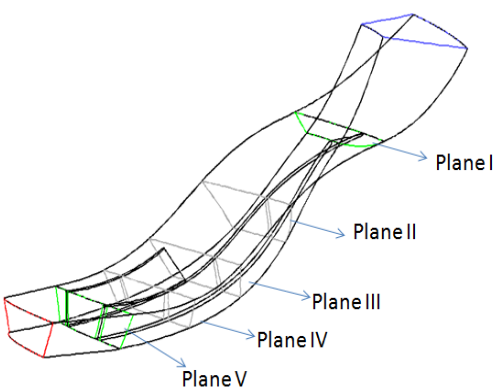


Figure-3 Meridional planes considered for relative velocity contours.

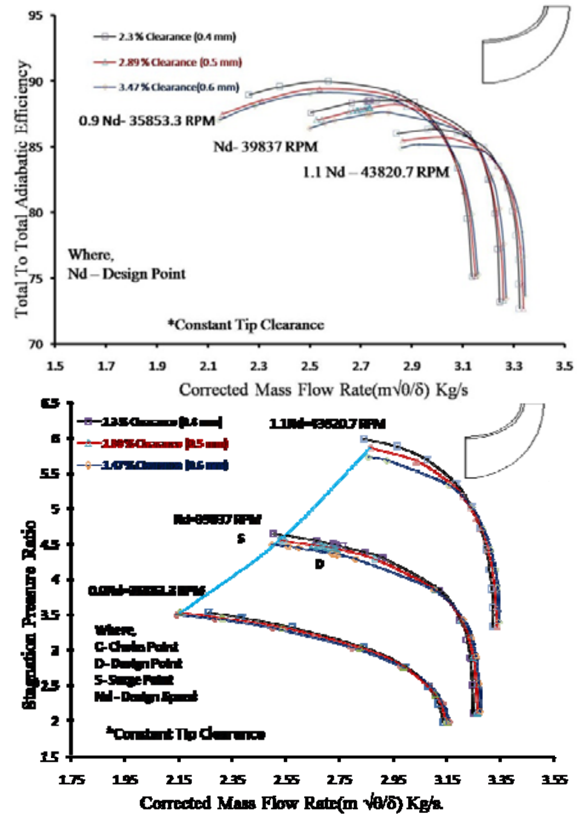


Figure-4 performance for constant tip clearance

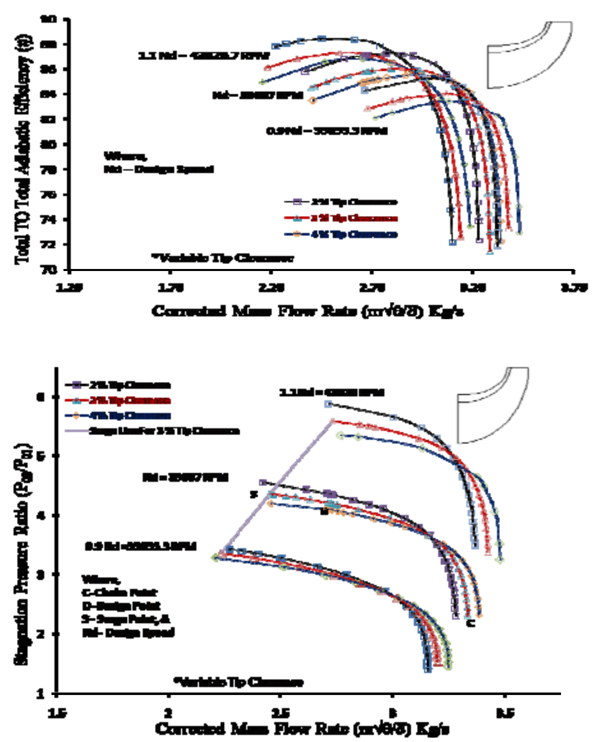


Figure-5 Performance for variable tip clearance

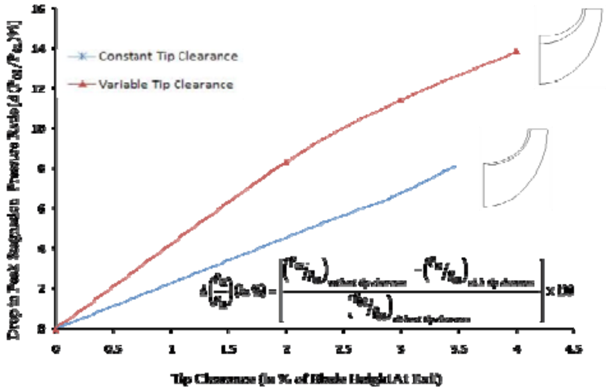


Figure-6 Peak stagnation pressure ratio drop.

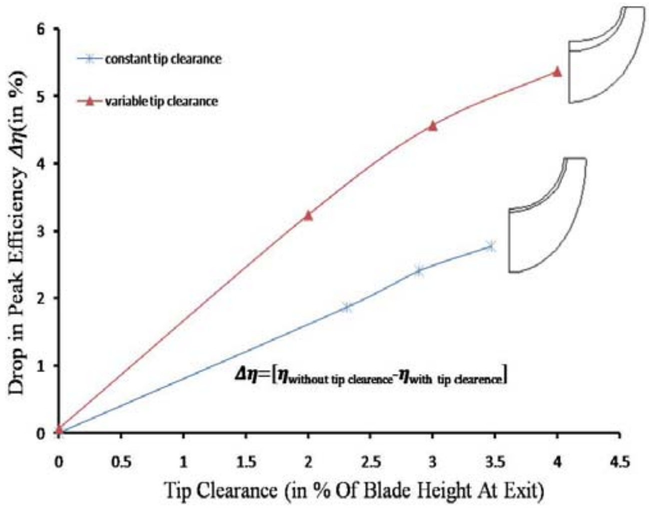


Figure-7 Peak efficiency drop.

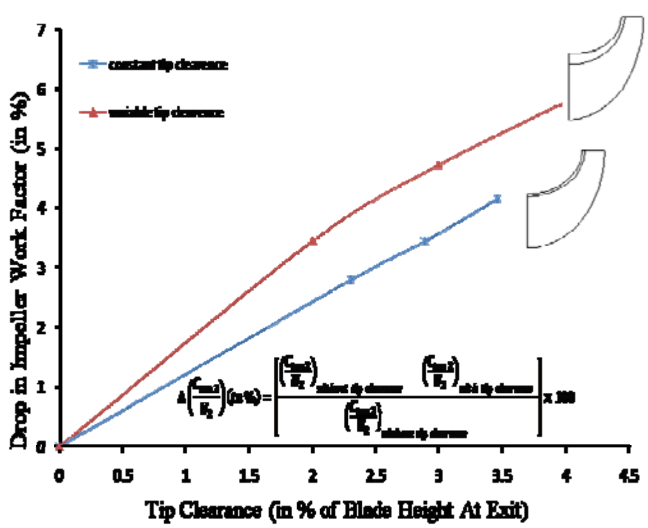


Figure-8 Drop in impeller work factors.

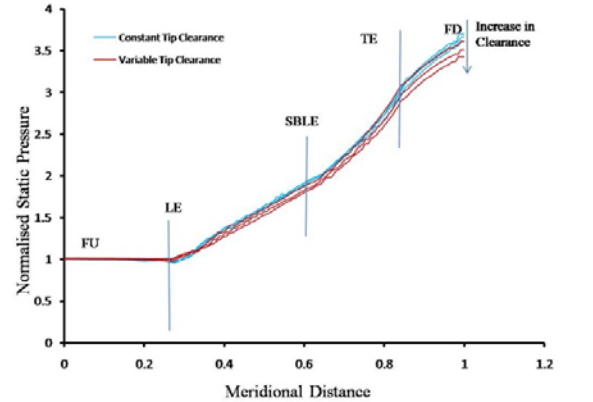


Figure-9 Static pressure distributions (near the shroud)

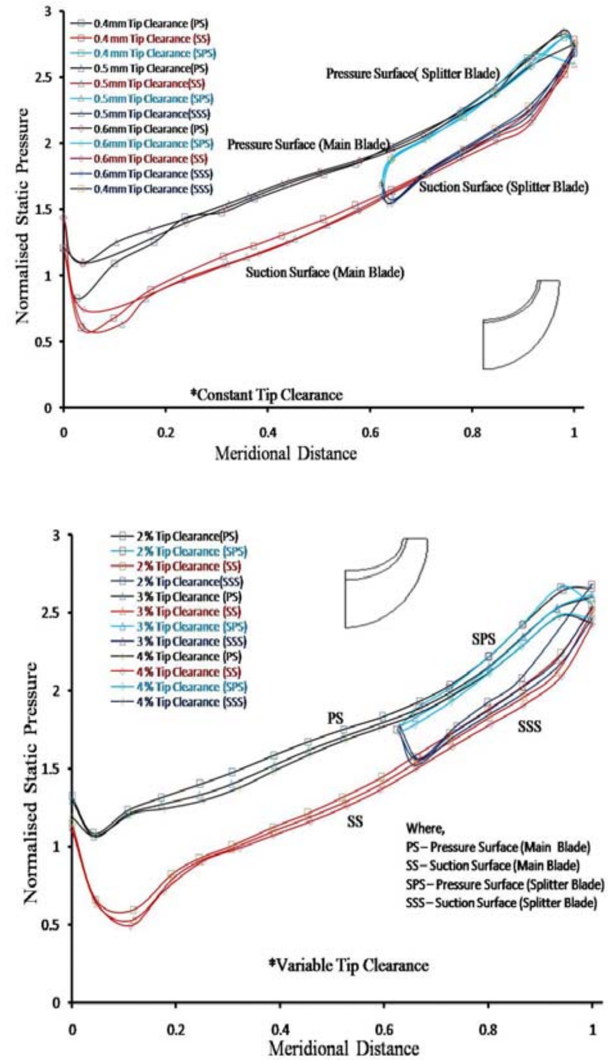
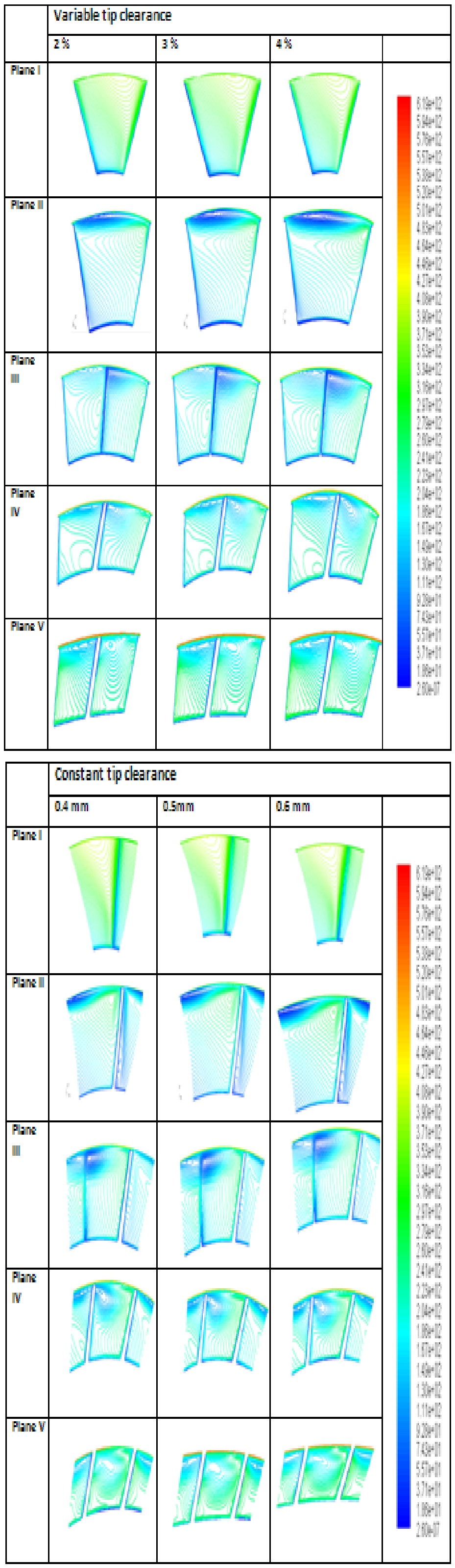


Figure-10 pressure distribution on blade surface (near the tip)



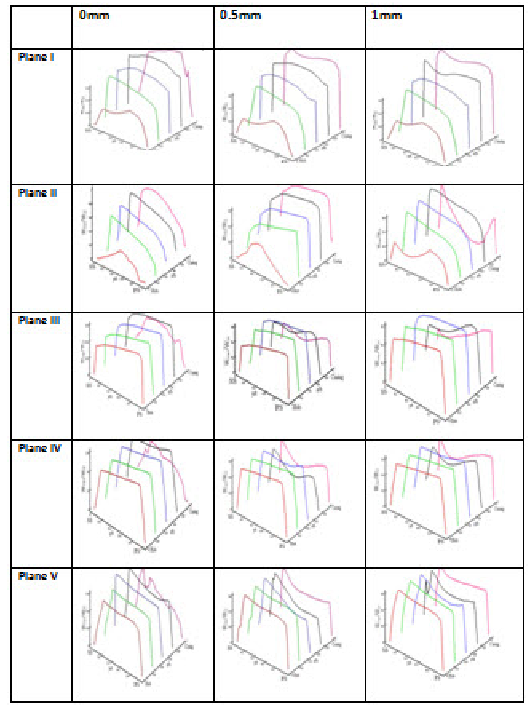


Figure-12 Relative velocity profile

Figure-11 Relative velocity contours